Study on dynamics of electro-hydraulic vibration device for vibratory experiments of steel piles in coral reefs

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Abstract: This paper presents a mathematical model of an electro-hydraulic vibratory exciter used for vibration testing of steel piles on coral reefs. Based on this model of electro-hydraulic vibrator, we have built characteristic curves to evaluate the stability of the system, the quality of the transition process, as well as parameters - amplitude, frequency. The research results are the scientific basis for design of electro-hydraulic vibratory exciter for vibration testing of steel piles in the coral foundation.
Keywords: Vibration exciter, electro-hydraulic vibrator, frequency.

1. Introduction

Vibration technology is an important basic technology, which has been widely used in the architectural engineering [1-3, 5], automotive industry [6], aerospace industry [7], and other related fields [4]. Vibration exciter is often designed to apply reciprocating motion on an object and produces vibrations. In general, there are three types of vibration exciters used in industrial applications: mechanical, electromagnetic, and electro-hydraulic. These types of vibration exciters are different in terms of the mechanism producing vibrations: mechanical vibrations by centrifugal force [8], electromagnetic vibrations by Lorenz force [9] and electrohydraulic vibrations by hydraulic pressure [10]. Due to its advantages of high power density, large output force, convenient operation, and load adaptive ability, electro-hydraulic vibratory exciter shows a broad application in compared with the mechanical or electromagnetic ones.

To study the load capacity of steel works of pile foundations on coral reefs when building marine structures, especially the ability to withstand the cyclic forces of sea waves, need to design a device causing vibrations to simulate the impact of ocean waves on steel piles. From the above purpose, in this paper, we focus on building a mathematical model of the device for building the dynamic characteristics and calculating frequency characteristics-amplitude which is scientific basis for designing devices using electro-hydraulic vibration.

2. Mathematical model of electro-hydraulic vibrator

To begin with, we have considered the principle scheme of the electro-hydraulic vibrator (see Fig. 1). The electrohydraulic vibration exciter consists mainly of a servo valve and a cylinder. By supplying a cyclically varied current to the electrical-to-mechanical transformer of the servo valve, the spool is excited into a reciprocating motion, which in turn varies flow to the chambers of the cylinder. This results in a vibration created by the piston of the cylinder and connecting loads.

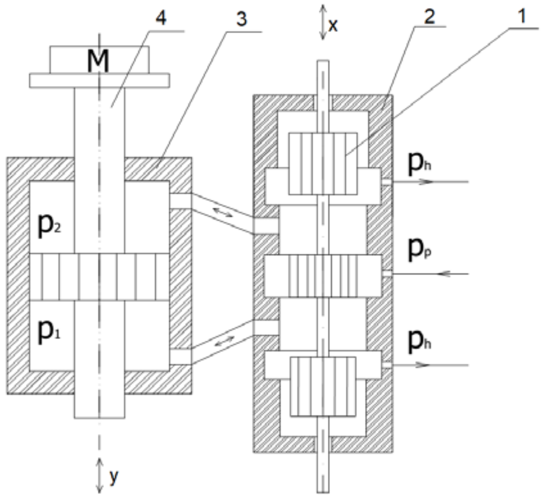

Fig. 1. Principle scheme of the electrohydraulic vibrator:
1 – spool sleeve; 2 – valve body; 3 – hydraulic cylinder; 4 – rod-piston; p_1- the lower chamber pressure; p_2- the upper chamber pressure; p_p –the pump pressure; p_h –the pressure in the return line; x – the displacement of the distribution valve slider; y - the displacement of plunger; M – the external forces;

Before the derivation of the mathematical model, simplifying assumptions are made: Simplifying assumption on the load - the force acts from the side of the control object to the executive part; Simplifying assumption regarding a source of an unlimited power supply; Simplifying assumption on the electric valve - processes related to the electric valve are not considered; Simplifying assumption regarding the throttling ideal hydro-distributor (micro-geometry, leakages, overlapping of working windows, gaps between the spool and bushing do not take into account);

Simplifying assumption regarding the drive - there are no fluid leaks, i.e. consider the drive to be sealed; Simplifying assumption connective turbo-wires - do not consider the hydraulic resistance of turbo-wires and wave processes in the turbo-wires, including their short.

It must also consider in this case that the electro-hydraulic vibrator without feedback, i.e. open loop.

2.1. Mathematical modeling of hydraulic mechanical part

Mathematical model concludes the hydraulic link and the mechanical link in the hydro-mechanical system is shown in Fig. 2.

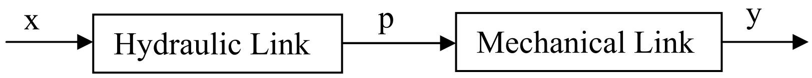

Fig. 2. The hydraulic link and the mechanical link in the hydro-mechanical system

It is customary to call the equations describing the hydraulic link - the equations of expense, and the equations describing the mechanical link - by the load equations.

For each working cavity, equations are compiled representing the balance of the mass expense:

$$Q^{M}_{\mathrm{in},i} - Q^{M}_{\mathrm{out},i} = Q^{M}_{\mathrm{cav},i} \qquad (1)$$

where $Q^{M}_{\mathrm{in},i}$ - the mass expense flowing into the cavity; $Q^{M}_{\mathrm{out},i}$ - the mass expense flowing out of the cavity; $Q^{M}_{\mathrm{cav},i}$ - the mass expense of the working environment.

Now we turn to the right part of equation (1):

$$Q^{M}_{\mathrm{cav},i} = \frac{dM_i}{dt} = \frac{d(\rho_i V_i)}{dt} = \rho_i \frac{dV_i}{dt} + V_i \frac{d\rho_i}{dt} \qquad (2)$$

where V_i – the volume of working fluid, $V_i = F(l_0 \pm y)$; F - the sectional area of the working cavity; l_0 - the cavity length at the neutral position of the piston; M_i – the volume of working fluid; ρ_i - the specific weight of fluid; p_i – the pressure; i – the cavity order.

The relationship between specific weight of fluid and pressure is:

$$\rho_i = \rho(p_i) \tag{3}$$

Taking the time derivative of equation (3) yields:

$$\frac{d\rho_i}{dt} = \frac{d\rho_i}{dp_i} \cdot \frac{dp_i}{dt} \tag{4}$$

Also, the compressibility factor of the working environment $\beta_{com,i} = \frac{dV_i}{V_i dp_i} = \frac{d\rho_i}{\rho_i dp_i}$ and the the elastic module of the working fluid $E_i = E(p_i) = \frac{1}{\beta_{com,i}} = \frac{\rho_i dp_i}{d\rho_i}$, therefore:

$$\frac{d\rho_i}{dt} = \frac{\rho_i}{E_i}\frac{dp_i}{dt} \tag{5}$$

The Eq.(3) ÷ Eq.(5) is substituted into Eq.(2) which yields:

$$Q^{м}_{\text{cav},i} = \pm\rho_i F\frac{dy}{dt} + \rho_i\frac{V_i}{E_i}\frac{dp_i}{dt} = \rho_i\left(\pm F\frac{dy}{dt} + K_{com,i}\frac{dp_i}{dt}\right) = \rho_i Q_{\text{cav},i} \tag{6}$$

where $Q_{\text{cav},i}$ the volumetric expense of the working environment; $K_{com,i}$ the compression factor of the working fluid.

$$Q_{\text{cav},i} = \pm Q_h + Q_{com,i} \tag{7}$$

where $Q_h = F\frac{dy}{dt}$ the geometric expense (ensures the movement of the piston); $Q_{com,i} = K_{com,i}\frac{dp_i}{dt}$ the compressibility expense (takes into account the compressibility of the working environment).

Now we turn to the left part of the equation (1):

$$Q^{м}_{\text{in},i} = \rho_i Q_{\text{in},i}\,;\; Q^{м}_{\text{out},i} = \rho_i Q_{\text{out},i}\,;\; Q^{м}_{\text{cav},i} = \rho_i Q_{\text{cav},i} \tag{8}$$

The Eq.(7) and Eq.(8) is substituted into Eq.(1) which yields:

$$Q_{\text{in},i} - Q_{\text{out},i} = Q_{\text{cav},i} = \pm Q_h + Q_{com,i} \tag{9}$$

Now we can write the equation of the expense for each working cavity assuming that:

$$\rho_i = \rho\left(1 + \frac{p_i}{E_i}\right) \approx \rho \text{ , as } \frac{p_i}{E_i} << 1 \tag{10}$$

The cavity 1 when x > 0:

$$Q_{sp,1} - Q_0 = Q_h + Q_{com,1} \tag{11}$$

$$\Rightarrow \mu_{sp} f_{sp} \sqrt{\frac{2}{\rho}\left(p_p - p_1\right)} = F\frac{dy}{dt} + K_{com,1}\frac{dp_1}{dt} + K_0(p_1 - p_2) \tag{12}$$

The cavity 2 when x > 0:

$$Q_0 - Q_{sp,2} = -Q_h + Q_{com,2} \tag{13}$$

$$\Rightarrow \mu_{sp} f_{sp} \sqrt{\frac{2}{\rho}\left(p_2 - p_h\right)} = F\frac{dy}{dt} - K_{com,2}\frac{dp_2}{dt} + K_0(p_1 - p_2) \tag{14}$$

where μ_{sp} the flow coefficient; f_{sp} - the cross section; K_0 – the leakage coefficient.

It is necessary to combine two Eq. (12) and Eq. (14), going to general $p = p_1 - p_2$. Since the valve is assumed ideal and symmetrical, then:

$$Q_{sp,1} = Q_{sp,2} \rightarrow p_p - p_1 = p_2 - p_h \tag{15}$$

Then we have a system of equations, from which p_1 and p_2 could be estimated:

$$\begin{cases} p_1 + p_2 = p_p + p_h \\ p_1 - p_2 = p \end{cases} \Leftrightarrow \begin{cases} p_1 = (p_p + p_h + p)/2 \\ p_2 = (p_p + p_h - p)/2 \end{cases} \tag{16}$$

Therefore:

$$p_p - p_1 = p_p - \frac{p_p + p_h + p}{2} = \frac{p_p - p_h - p}{2} = \frac{p_n - p}{2} \tag{17}$$

$$p_2 - p_h = \frac{p_p + p_h - p}{2} - p_h = \frac{p_p - p_h - p}{2} = \frac{p_n - p}{2} \tag{18}$$

where $p_n = p_p - p_h$ the supply pressure.

We also consider the coefficient of compressibility of a liquid $K_{com,i}$:

$$K_{com,1} = \frac{V_1}{E_1};\ K_{com,2} = \frac{V_2}{E_2}$$

To combine equations (12) and (14) we assume that:

$$K_{com,1} \approx K_{com,2} = K_{com} = \frac{V}{E} = \frac{Fl}{E},\ E_1 \approx E_2 = E,\ V_1 \approx V_2 = V \tag{19}$$

From Eq.(12) and Eq.(14), we can obtain the following expression:

$$\mu_{sp} f_{sp} \sqrt{\frac{1}{\rho}\left(p_n - p\right)} = F\frac{dy}{dt} + \frac{K_{com}}{2}\frac{dp}{dt} + K_0 p \tag{20}$$

where $f_{sp} = \pi d_{sp} x$ is the open area of the valve; d_{sp} is the diameter of the slider.

$$\Rightarrow \mu_{sp} \pi d_{sp} x \sqrt{\frac{1}{\rho}\left(p_n - p\right)} = F\frac{dy}{dt} + \frac{K_{com}}{2}\frac{dp}{dt} + K_0 p \tag{21}$$

The equation (21) – is the combined equation of the drive expense for positive displacement of the valve (x > 0). It is necessary to check its reversibility.

When $x > 0 \rightarrow p_1 > p_2 \rightarrow p > 0$ - positive displacement

When $x < 0 \rightarrow p_1 < p_2 \rightarrow p < 0$ - negative displacement

With negative displacement, the discharge and drain edges change places. As a result of the similar calculations, we obtain the combined equation of the drive expense for positive displacement of the valve ($x < 0$):

$$\mu_{sp} \pi d_{sp} x \sqrt{\frac{1}{\rho}\left(p_n + p\right)} = F\frac{dy}{dt} + \frac{K_{com}}{2}\frac{dp}{dt} + K_0 p \tag{22}$$

Now we combine equations (21) and (22), for this we set:

$$\text{sign (p)} = \begin{cases} +1 & \text{when} \quad \text{p} > 0 \\ -1 & \text{when} \quad \text{p} < 0 \end{cases}$$

and we obtain the follow combined equation of the drive expense:

$$\mu_{sp} \pi d_{sp} x \sqrt{\frac{1}{\rho}\left(p_n - p \cdot sign(p)\right)} = F\frac{dy}{dt} + \frac{K_{com}}{2}\frac{dp}{dt} + K_0 p \tag{23}$$

The load equation is obtained from the equation of mechanics (2-nd Newton's law):

$$M\frac{d^2 y}{dt^2} = \sum P_i = P_{dr} - P_{res} \tag{24}$$

where M - the mass of moving parts; $P_{dr} = F \cdot p$ - the driving force;

$$P_{res} = P_{fri} + G = K_{fri}\frac{dy}{dt} + G$$

P_{fri} - the friction force; K_{fri} - the coefficient of the viscose friction; G – the resistance force;

Then the load equation has the form:

$$M\frac{d^2y}{dt^2}=F\cdot p-K_{fri}\frac{dy}{dt}-G \tag{25}$$

Thus, the work of the hydraulic is estimated by the following equations:

The combined equation of the expense:

$$\mu_{sp}\pi d_{sp}x\sqrt{\frac{1}{\rho}\left(p_n-p\cdot sign(p)\right)}=F\frac{dy}{dt}+\frac{K_{com}}{2}\frac{dp}{dt}+K_0p \tag{26}$$

The load equation:

$$M\frac{d^2y}{dt^2}=F\cdot p-K_{fri}\frac{dy}{dt}-G \tag{27}$$

2.2. Mathematical modeling of electric valve part

Electric valve is an electromechanical system, therefore in the mathematical description it is necessary to obtain differential equations for the electronic circuit (control windings) and for the mechanical system – the moving part of the electric valve.

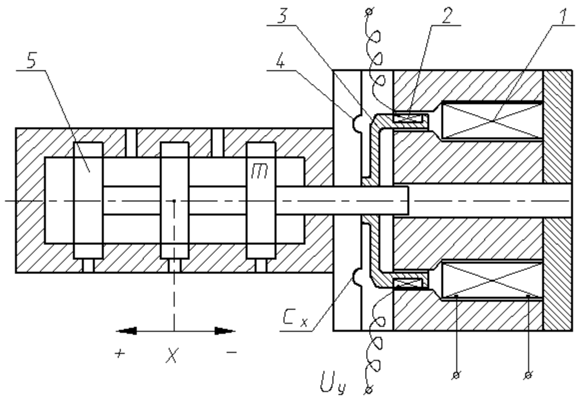

Fig. 3. The scheme of principle of electric valve parts

1 – bias coil; 2 – control coil; 3 – frame; 4 – elastic suspension; 5 – valve.

Derivation of differential equations is performed under the following basic assumptions: There is no hysteresis phenomenon in the metal of the magnet wire; The displacement of the rod of the electric valve is small; There is no dry friction. Fulfillment of these requirements allows us to obtain a linear mathematical model of the electric valve. The scheme of the principle of the electric valve parts is shown in Fig. 3.

In the translational motion of the moving parts of electric valve, the motion equation has the form:

$$m\frac{d^2x_{sp}}{dt^2}=F_{dr}-F_{fri}-F_{sus} \tag{28}$$

where m - the mass of the moving parts; $F_{dr}=B_v l_y i_y=K_{x,i}i_y$ – the driving force of valve; $K_{x,i}$ - the coefficient of the driving force; $F_{fri}=h_x\frac{dx_{sp}}{dt}$ - the force of viscous friction applied to the valve; h_x - the coefficient of viscose friction; $F_{sus}=c_x x_{sp}$ – the force, acting from the side of the elastic suspension of electric valve; c_x - the suspension stiffness.

The second differential equation is the equation of the electrical circuit of the control coil:

$$U_y=U_L+U_R+e_x \tag{29}$$

where $U_L=L\frac{di_y}{dt}$ – the voltage drop on the inductive resistance of the control coil; L - the inductance of the control coil; $U_R=Ri_y$ – the voltage drop on the active resistance of the control coil; R- the active resistance of the control coil; $e_x=Bl\frac{dx}{dt}=K_{x,u}\frac{dx}{dt}$ – arising from the movement of the conductor of the control coil in the magnetic field of the bias coil – induction of electric valve; $K_{x,u}$ - the anti-coefficient of electric valve.

It should be noted that anti-coefficient is equal to the coefficient of the driving force:

$$K_{x,u}=K_{x,i} \tag{30}$$

After disclosure (discussion) of the meaning and content of the components in equations (28) and (29), we obtain a system of differential equations describing the dynamics of electric valve as:

$$m\frac{d^2x_{sp}}{dt^2}+h_x\frac{dx}{dt}+c_xx_{sp}=K_{x,i}i_y \quad (31)$$

$$L\frac{di_y}{dt}+Ri_y=U_y-K_{x,u}\frac{dx_{sp}}{dt} \quad (32)$$

3. Simulation analysis

In order to demonstrate the above model, the following input parameters are used: Piston diameter of cylinder: D = 50 mm; rod diameter of cylinder: d = 35 mm; displacement of piston: l_0 = 0,25 m; viscous friction coefficient: K_{fri} = 10^4 (Ns/m); dynamic viscosity: μ_{sp} = 0,65; Diameter of valve spool: d_{sp} = 5 mm; pump flow: Q_{max} = 5 l/min; pressure: p_{max} = 200 bar; voltage on electric valve: U_y = 24V.

With the above input parameters of the device, we have surveyed and received the results of the dynamic parameters of the device causing vibration in no-load mode (Fig. 4) and the mode of causing steel pile vibration (Fig. 5) at 1 Hz frequency.

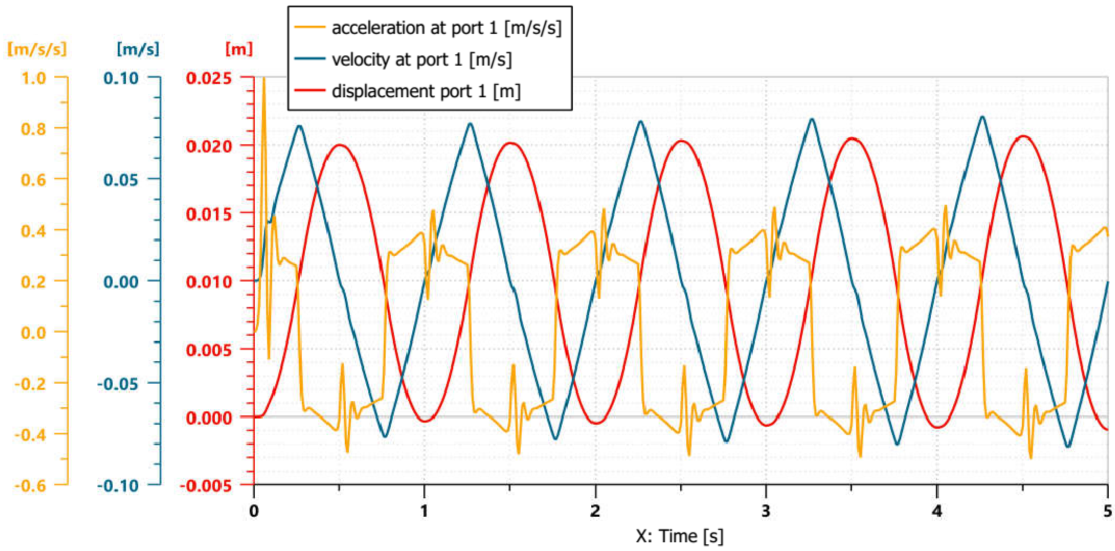

Fig. 4. The kinetic parameters of electro-hydraulic vibratory exciter in no-load mode at 1Hz frequency

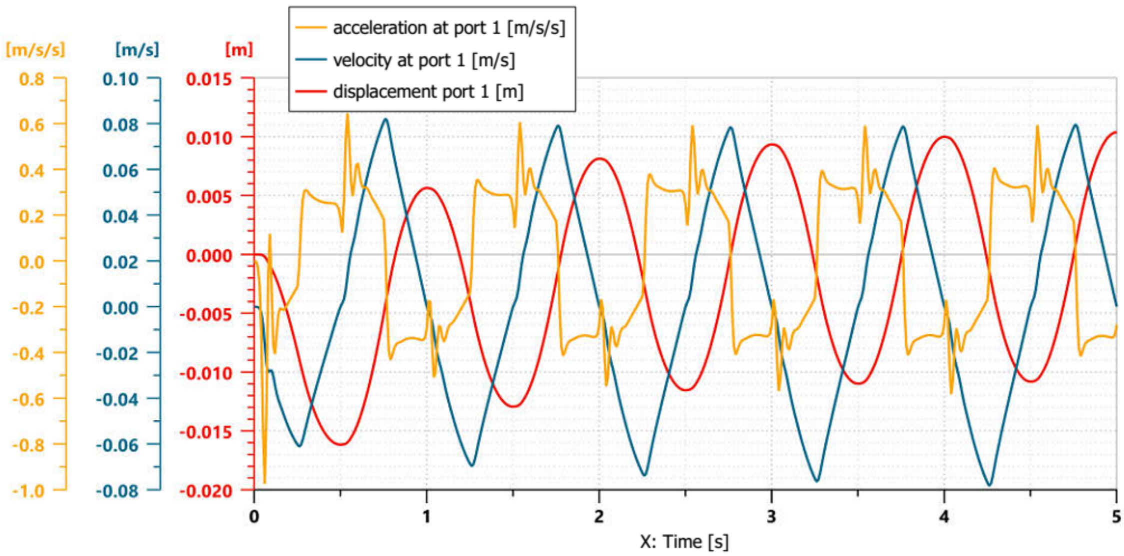

Fig. 5. The dynamic parameters of electro-hydraulic vibratory exciter in the mode of causing steel pile vibration at 1Hz frequency

The frequency-amplitude characteristic of the electro-hydraulic vibratory exciter is shown in Figure 6.

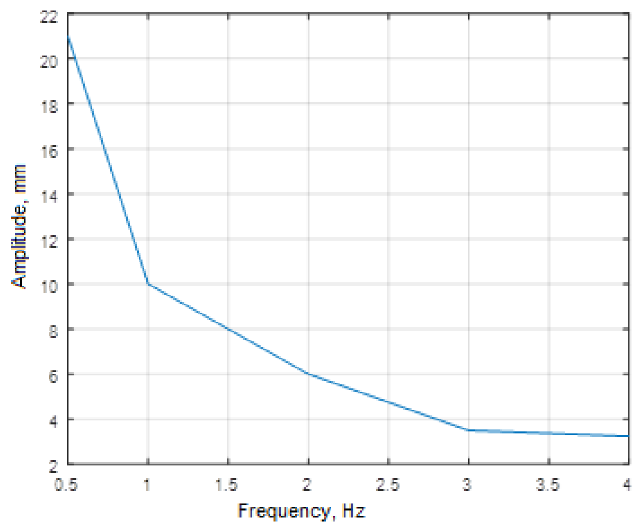

Fig. 6. The frequency-amplitude characteristic of the electro-hydraulic vibratory exciter

From the obtained results about the dynamic parameters of the electro-hydraulic vibratory exciter, the following remarks can be stated:

- As shown in Figures 4 and 5, the change of acceleration in both load and non-load cases is relatively large. Such change of acceleration creates large inertial forces, which adversely affects the system. Therefore, in the hydraulic system of this device, it is compulsory to have a hydraulic battery to absorb oscillating pulses;

- As shown in Figure 5, when vibrating steel piles in the coral reef, which is a slippery environment, the oscillation center movement will occur, resulting in unsatisfactory vibration mode. Therefore, a position sensor to switch the circuit to aim at fixing the oscillating should be used;

- According to Figure 6, when the pump flow rate is constant at 5 l/min, the amplitude of oscillation decreases with increasing of the oscillation frequency. To ensure that the required amplitude value is always at 10 mm when the frequency of oscillation increases, it is essential to increase the pump flow level at the corresponding level.

4. Conclusions

In this paper, the mathematical model of the electro-hydraulic vibratory exciter was used in the experiments that produces vibration for installing steel piles on the coral reefs. The dynamic parameters and frequency characteristics of the electro-hydraulic vibratory exciter were investigated. The obtained results in this work are the scientific basis for designing devices using electro-hydraulic vibration for piling testing of steel piles in the coral reefs.

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